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THE ANALYTICAL DESIGN OF A TURBO-JET COMBUSTION CHAMBER

A Thesis Submitted to the Graduate Faculty
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I. SUMMARY

A "can" type combustion chamber was designed using all available proved concepts. A survey of existing designs was made to arrive at a logical design point. The fact that parallel flow would exist within the chamber was noted, and the burner was designed to obtain the required flow with an equal pressure drop across all the parallel paths. The holes for introducing the air into the burner basket were placed in such a manner that the total area into the burner basket, plotted against distance downstream, fell on an arc of a circle. This made the design follow several proved concepts automatically.

The performance was calculated and compared with existing designs. The comparison was quite favorable, with the exception of combustion efficiency. Since the empirical equation for combustion efficiency was obtained from tests under slightly different conditions than the design point, doubt exists as to the validity of the values obtained by its use.

II. INTRODUCTION

Many combustion chambers for turbo-jet engines have been built and tested; however, they all fall into one of two classifications. The most common type is the tube or "can" combustion chamber. The other is the annular type.

The "can" type of combustion chamber, as its name implies, is a metal cylinder or can with an inner liner or "basket" which separates the air into two parts. The early turbo-jet engines designed by Air Commodore F. Whittle of Great Britain used "can" combustion chambers of the "return flow" type. In this type of chamber, the air passes over the entire length of the combustion zone before entering to be mixed with the fuel and burned. Thus the air is preheated by what would otherwise be waste heat. This advantage is more than overcome by the disadvantage of a high pressure loss. The more modern "can" type combustion chambers are of the "straight through" variety, i.e., there are no turns encountered within the chamber itself.

The annular type of combustion chamber has the distinct advantage of offering less frontal area for a given capacity than a battery of "cans" of equal capacity. The annular combustion chamber has the disadvantage of having to be tested in its entirety, while one "can" of a multi-can battery can be tested. Work is being done at Westinghouse in testing a small segment of an annular chamber and using the results to predict the performance of the complete unit.

No results have been published, but it is expected that this method of testing will prove satisfactory.

Since the "can" type combustion chamber is more easily built and the testing procedures for the chamber are more nearly standardized, it was decided to base the analytical design on this type of combustion chamber.

III. PURPOSE OF INVESTIGATION

"Cut and try" has been the method employed in the past in designing combustion chambers for turbo-jet engines. This is a long and costly process, and luck plays a considerable part in obtaining results. It is the purpose of this investigation to design a combustion chamber using analytical methods. Proved concepts will be used where they are available. It is hoped that this procedure will produce a satisfactory chamber with little or no alteration necessary.

IV. PROBLEMS INVOLVED IN THE DESIGN OF A COMBUSTION CHAMBER

The function of a turbo-jet combustion chamber is to heat the air from the compressor to the temperature required at the entrance to the turbine. This temperature is usually taken to be 1500° F, this being the maximum temperature at which present day turbine wheels can operate. Sufficient fuel to achieve this temperature gives a fuel-air ratio far too lean for satisfactory combustion. Therefore, the air is divided into primary air and secondary air, the primary air being just sufficient to produce a stoichiometric fuel-air mixture with the required fuel and the secondary air being introduced to cool the products of combustion before reaching the turbine.

The primary air must be introduced into the combustion chamber with enough turbulence to insure good mixing with the fuel. This turbulence also speeds the burning and gives a stable flame.

The secondary air must be brought in with enough large-scale turbulence to complete the burning and to mix thoroughly with the products of combustion. Any stratification due to incomplete mixing will result in layers of air at temperatures higher than the allowable reaching the turbine. Hot spots will be caused, and a failure of the metal may occur.

High turbulence and a long combustion chamber tend to give more complete burning and better mixing of secondary

air and combustion products. Unfortunately, these characteristics are not compatible with an efficient design. High turbulence produces high pressure losses through the combustion chamber. Since compressor efficiencies and pressure ratios are critical, the additional pressure loss which results from high turbulence must be avoided. Therefore, just enough turbulence to complete the mixing should be used. Anything in excess of this, while insurance against turbine hot spots, cuts down the combustion chamber efficiency.

Since turbo-jet engines are used on airplanes, space and weight are at a premium. This precludes the use of an excessively long combustion chamber to complete the mixing. Lengths of three to four diameters are current practice.

The rate at which the secondary air is mixed with the combustion products is of great importance. Too much air introduced too soon may chill the flame front sufficiently to prevent the combustion reaction from going to completion. This, of course, would reduce the combustion efficiency. Reference 1 states that the first secondary air should not be introduced nearer than six to eight inches downstream from the fuel nozzle.

Experiments conducted at the California Institute of Technology show that the shape of the holes used to let the air through the inner liner has an effect on the pressure drop through these holes and also an effect on the mixing of the air with the combustion products. Figures 1 and 2 show the effects of shape on mixing. A thin slot parallel to the

flow gives the best mixing, a square or circular hole next best, and a slot perpendicular to the flow the worst.

The square hole and the slot are undesirable, however, because high stress concentrations would be produced at the sharp corners.

The round hole has found almost universal acceptance. The use of a bell mouth on these holes reduces the pressure drop across the holes by nearly thirty per cent without affecting the mixing.

These experiments also showed that arranging the holes in lines with one hole directly downstream from the other gave the greatest amount of turbulence for a given pressure drop.

Since there are no facilities available for testing the altitude performance of a combustion chamber, no attempt will be made to take this requirement into consideration.

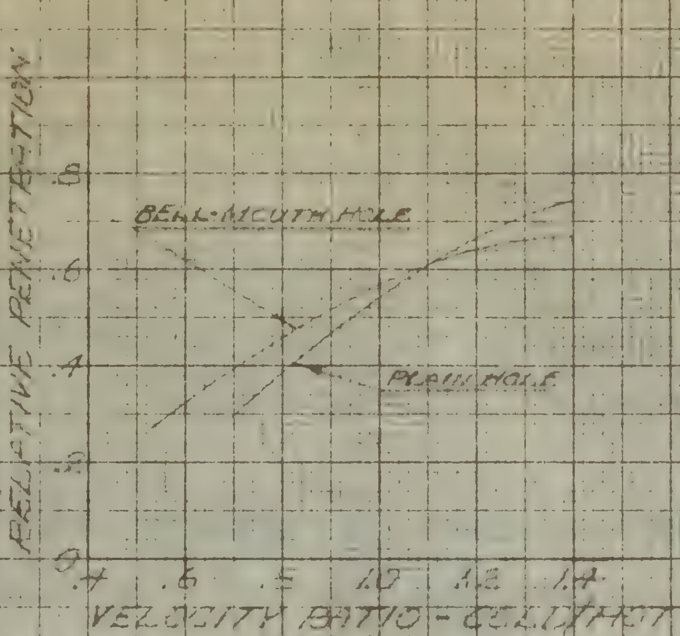

FIGURE 1
EFFECT OF HOLE MOUTH
SHAPE ON MIXING.
(FROM REF 1)

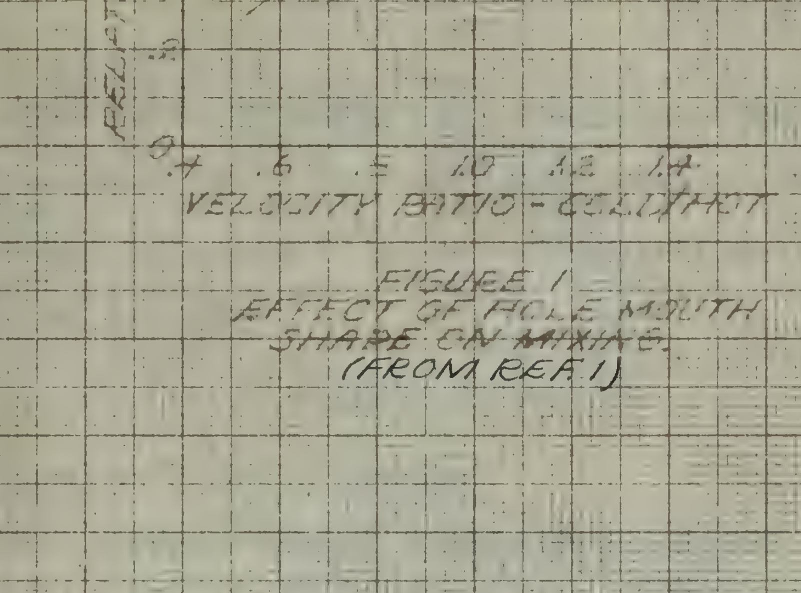

FIGURE 2
EFFECT OF HOLE SHAPE
ON MIXING.
(FROM REF 1)

V. SELECTION OF THE DESIGN POINT

In order to arrive at a suitable design point, a survey of several current model turbo-jet engines was made. Table I lists the design factors for the combustion chambers of these engines.

The "loading factor" or "combustion intensity factor," as it is sometimes called, is a parameter originated by the British for comparing different combustion chambers. It is defined as follows:

$$I = \frac{q}{PV}$$

where q = heat input in Btu/hr

P = pressure inside chamber in atmospheres

V = volume available for combustion inside basket in ft^3

I = loading factor in Btu/hr-ft^3-atm.

The heat input is computed by multiplying the fuel flow rate in pounds per hour by the lower heating value of the fuel. The volume available for combustion inside the basket is taken from the fuel injection nozzle to the turbine entrance.

It can readily be seen that the combustion intensity factor is a good indication of the conditions under which combustion takes place. For instance, for a given heat input, the larger the volume available for combustion, the greater will be the amount of air available to absorb the heat released and, consequently, the lower is the intensity

factor. Again, for a given heat input and a given volume available for combustion, the lower the pressure, the smaller will be the mass of air and the higher is the temperature rise from combustion; hence, the combustion intensity factor will be higher.

Table I. Design Factors for Combustion Chambers of Current Model Turbo-Jet Engines.*

Engine	N	v_r	I	D	l/D	L/D
TG-180 (J-35)	8	100	3.1×10^6	7.3	3.35	4.80
I-40 (J-33)	14	130	4.6×10^6	5.5	3.80	4.80
Nene	9	80	2.0×10^6	10.0	2.20	2.70
Derwent V	9	80	2.0×10^6	10.0	2.60	3.30
Goblin II	16	100	3.5×10^6	6.0	3.80	4.50
19XB	Annular	-	9.0×10^6	-	-	-
9.5B	Annular	-	10.0×10^6	-	-	-

* I - Loading factor or combustion intensity factor, Btu per hour per cubic foot per atmosphere, based on total volume available for combustion and mixing.

N - Number of combustion chambers.

l - Length in inches from fuel nozzle to end of basket.

L - Length in inches from fuel nozzle to turbine entrance.

v_r - Reference velocity in feet per second, computed using maximum cross-sectional area of combustion chamber.

D - Diameter of basket in inches.

The British have tried to correlate this combustion intensity factor with combustion losses. While no close

correlation exists between different combustion chambers, curves of individual chambers show that combustion losses increase rapidly with increasing combustion intensity factor. A typical graph of combustion losses vs. combustion intensity factor is presented in Figure 3 for the Lucas B-37 chamber. However, these losses may be inordinately high since, at sea level, values of combustion intensity factor as high as 9×10^6 are possible with a combustion efficiency of 95 per cent (Ref. 2).

The reference velocity, v_r, is another British parameter. It is used to get a rough check on the total losses through the combustion chamber. Reference 3 shows that individual combustion chambers have a pressure loss of from 20 to 30 times the velocity head computed for the reference velocity. Thus it can be seen that a low value of reference velocity should be used in designing a combustion chamber.

Some of the design conditions were dictated by the air source which would be available to test the combustion chamber when built. This source is from the second stage of an Allison V-1710 supercharger and is about three pounds per second at 1.6 atmospheres and 200° F.

As a result of this survey and from consideration of the air source available for testing, the following design point was selected:

Altitude = sea level

w = 3 lb/sec

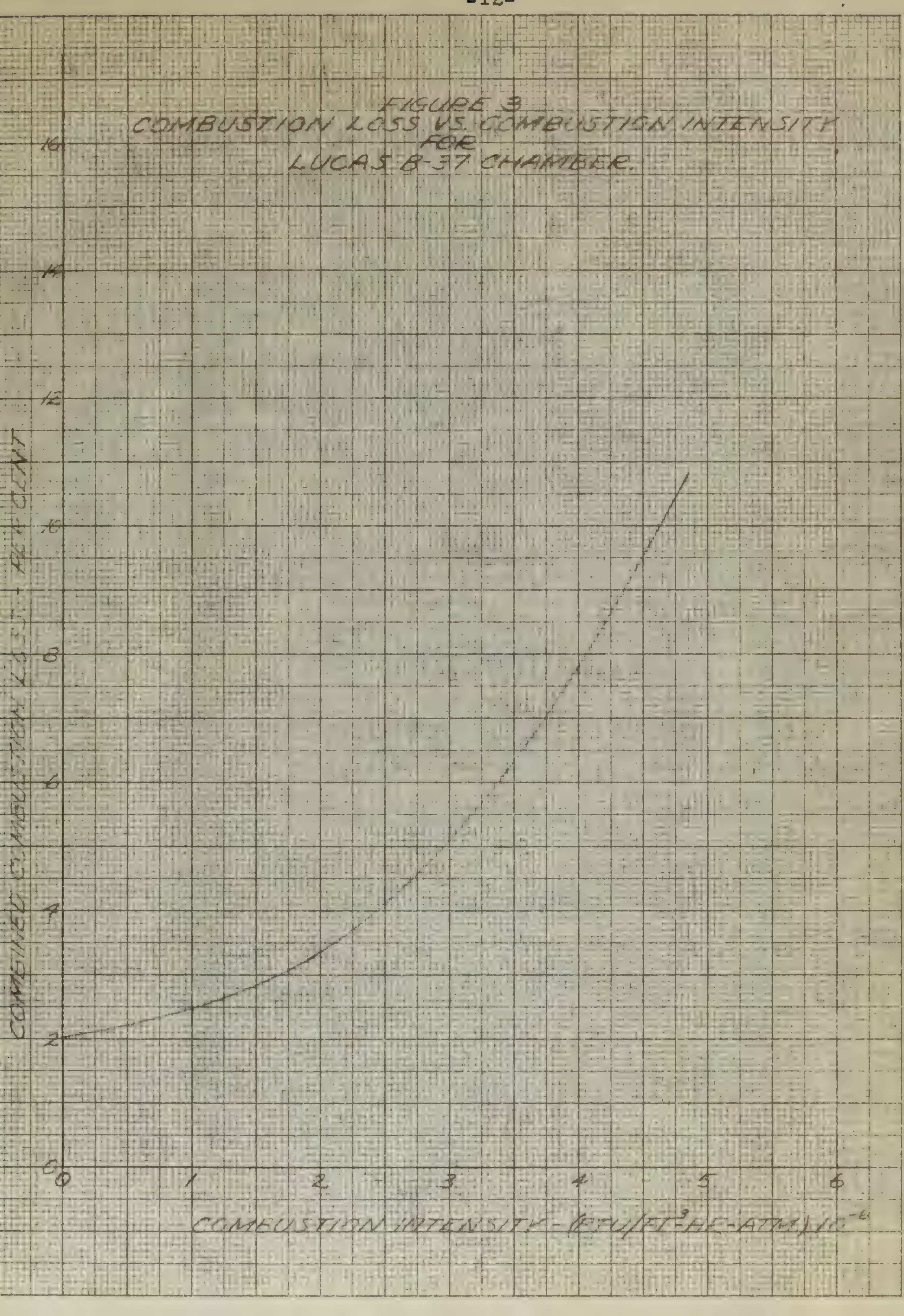

FIGURE 3
COMBUSTION LOSS VS. COMBUSTION INTENSITY
FOR
LUCAS B-37 CHAMBER.

P_{4T} = 1.6 atmospheres

T_{4T} = 200° F or 660° R

T_{5T} = 1500° F or 1960° R

V_r = 100 ft/sec

M_r = .08

L/D = between 3 and 4

l/D = between 2 and 3.

VI. DESIGN PROCEDURE AND CALCULATIONS

Maximum Diameter of Combustion Chamber

$$v_r = M_r(a)$$

where v_r = reference velocity in ft/sec

M_r = reference Mach number

a = speed of sound in ft/sec = $49\sqrt{T_{4T}}$

where T_{4T} = temperature in °R at entrance to combustion chamber

$$v_r = .08 \times 49\sqrt{660}$$

$$= 100 \text{ ft/sec}$$

$$\rho = \rho_{sc} \times \frac{P_{4T}}{P_{sc}} \times \frac{T_{sc}}{T_{4T}}$$

where ρ = density of air in lb/ft^3

ρ_{sc} = density of air in lb/ft^3 at standard conditions

P_{4T} = total pressure in lb/ft^2 at entrance to combustion chamber

P_{sc} = atmospheric pressure in lb/ft^2 at standard conditions

T_{sc} = temperature in °R at standard conditions

$$\rho = .07651 \times \frac{3380}{2115} \times \frac{520}{660}$$

$$= .0965 \text{ lb/ft}^3$$

$$A = \frac{W}{\rho v_r}$$

where A = area of combustion chamber in ft^2

w = air flow in lb/sec

$$A = \frac{3}{.0965 \times 100}$$

$$= .29 \ ft^2$$

$$D = \sqrt{\frac{A}{/4}}$$

where D_{max} = maximum diameter of combustion chamber in ft

$$D_{max} = \sqrt{\frac{.29}{.785}}$$

$$= .61 \ ft = 7.33 \ in$$

Fuel Required

$$c_{p_{ave}} = \frac{c_{p4} + c_{p5}}{2}$$

where $c_{p_{ave}}$ = average specific heat at constant pressure in Btu/lb air - °R

c_{p4} = specific heat at constant pressure in Btu/lb air - °R at entrance to combustion chamber

c_{p5} = specific heat at constant pressure in Btu/lb air - °R at exit from combustion chamber

$$c_{p_{ave}} = \frac{.2403 + .277}{2}$$

$$= .256 \ \text{Btu/lb air - °R}$$

$$q = c_{p_{ave}} (T_{5T} - T_{4T})$$

where q = heat required in Btu/lb air

T_{5T} = total temperature at combustion chamber exit in °R

$$q = .256(1960 - 660)$$

$$= 343 \text{ Btu/lb air}$$

$$w_f = \frac{qW}{\text{L.H.V.}}$$

where w_f = fuel required in lb/sec

L.H.V. = lower heating value of fuel in Btu/lb

$$w_f = \frac{343 \times 3}{18500}$$

$$= .056 \text{ lb fuel/sec}$$

Primary Air Required

In order to obtain sufficient combustion in the primary zone, there should be a stoichiometric mixture of fuel and air in this zone. Thus, the fuel-air ratio must be .067.

$$w = \frac{w_f}{.067}$$

where w = weight of primary air required in lb/sec

$$w = \frac{.056}{.067}$$

$$= .85 \text{ lb/sec}$$

Volume Available for Combustion

$$V = \frac{q}{FI}$$

where V = volume available for combustion in ft^3

q = heat input in Btu/hr

P = pressure inside chamber in atmospheres

I = combustion intensity factor in Btu/hr-ft^3-atm

$$V = \frac{.056 \times 3600 \times 18500}{1.6 \times 8 \times 10^6}$$

$$= .29 \text{ ft}^3$$

Configuration of Burner Basket

As shown in Figure 4, the burner is made up of three sections---hemisphere or dome, cylinder, and tail cone. Thus,

$$V_T = V_H + V_C + V_{TC}$$

where V_T = total volume in ft^3 = .29 ft^3

V_H = volume of hemisphere in ft^3

V_C = volume of cylinder in ft^3

V_{TC} = volume of tail cone in ft^3

Then,

$$V_T = \frac{\pi D^3}{12} + \frac{\pi D^2 l_C}{4} + \frac{h}{3}\left(\frac{\pi D^2}{4} + \frac{\pi d^2}{4} + \sqrt{\frac{\pi D^2}{4} \times \frac{\pi d^2}{4}}\right)$$

where D = diameter of cylinder in ft

d = smaller diameter of tail cone in ft

l_C = length of cylinder in ft

h = length of tail cone in ft

Assuming that D = 6 in = 1/2 ft, d = 4 in = 1/3 ft, and h = 5 in = 5/12 ft:

$$.29 = \frac{.785}{4} l_C + \frac{.262}{8} + \frac{5}{36}\left(\frac{.785}{4} + \frac{.785}{9} + \sqrt{\frac{.785}{4} \times \frac{.785}{9}}\right)$$

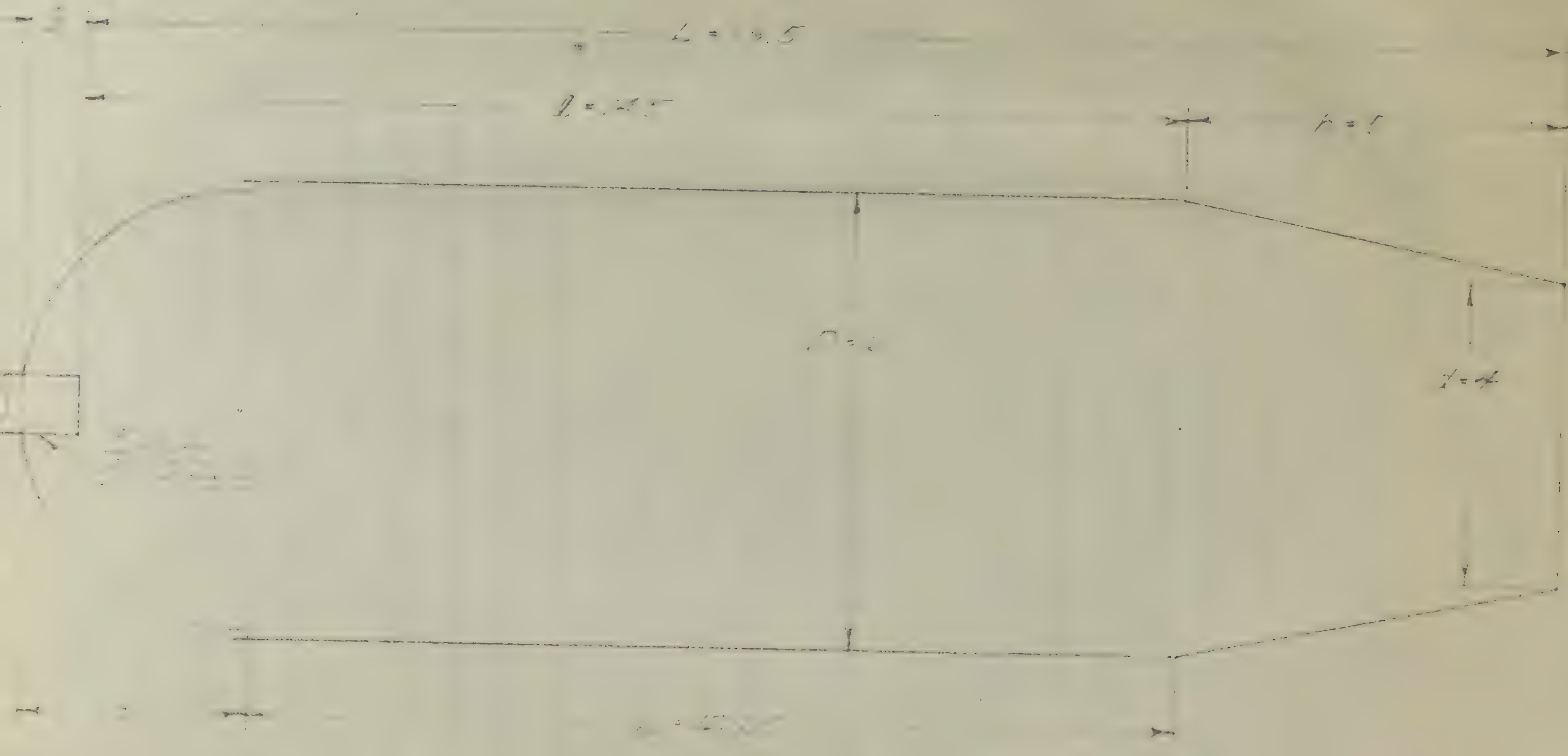

FIGURE 4 SKETCH OF BURNER BASKET, DOME, AND TAIL CONE.

Solving for l_c,

$$l_c = 1.02 \text{ ft} = 12.25 \text{ in}$$

Then,

$$l = l_c + 2.25$$

where l = length from fuel nozzle to downstream end of cylinder in inches

2.25 = length from fuel nozzle to upstream end of cylinder in inches

$$l = 12.25 + 2.25 = 14.5 \text{ in}$$

Also,

$$L = l + h$$

where L = length from fuel nozzle to end of tail cone in inches

$$L = 14.5 + 5 = 19.5 \text{ in}$$

Then,

$$\frac{l}{D} = 2.42 \text{ and } \frac{L}{D} = 3.25$$

Since the values of l/D and L/D are between the desired values of 2 and 3 and 2 to 4 respectively, the dimensions of the burner basket were selected as assumed and calculated in the foregoing procedure and may be summarized as follows:

$D = 6$ in

$d = 4$ in

$h = 5$ in

$l_c = 12.25$ in

$l = 14.5$ in

$L = 19.5$ in

Design of Primary Zone

The problem of introducing the primary air with the proper turbulence is the most important single factor in the design of a combustion chamber, according to A. J. Nerad (Ref. 4). When done properly, a smooth, continuous ignition process occurs in the dome end of the combustion chamber.

Best results have been obtained when everything possible has been done to produce a strong reverse flow in the primary zone. This reverse or "back-flow" produces an "ignition eddy" or "tore," as it is referred to in Nerad's paper.

To establish the existence of this "tore" beyond any doubt, various tests have been carried out with pressure probes, along with other tests such as injecting the fuel into the first ring of holes instead of at the nozzle. These tests have proved that the flow is essentially that which is shown in Figure 5.

Looking at the end view of Figure 5, it is obvious that the eight strong jets of air impacting at the center must produce a substantial axial flow both into and out of the plane of the paper. The axial flow which goes forward is deflected outward by the dome and emerges in the relatively unobstructed areas between the rows of air inlets. This air has mixed with fuel and been ignited and now is "hot burning gases." These gases pass in close proximity to the incoming cold air and part is entrained by it, re-

FIGURE 4. CUTAWAY DIAGRAM OF COMBUSTION CHAMBER SHOWING FLOW OF AIR

sulting in extremely fast preheating of the incoming combustion air. This has a very beneficial effect on good ignition and smooth combustion.

Since it is desirable to have this "back-flow" as strong as possible, all the primary air, except a small amount necessary for cooling and carbon prevention, will be introduced through holes in the basket downstream from the fuel nozzle.

The flow through the combustion chamber is parallel, i.e., the mass of air which enters the combustion chamber through any hole or slot in the burner basket does not in turn pass through any other hole or slot on its way to the turbine, and, likewise, the air which travels through the annular path in the tail cone has not previously passed through any hole or slot in the basket. Since the flow is parallel, the friction pressure drop through any part of the burner must equal that through any other part. This fact was used in designing the combustion chamber.

The design of the primary zone was completed as follows:

First, the weight flow through a ring of eight 1/2 inch primary holes located 4-1/2 inches from the nozzle was determined approximately by the formula:

$$w = \rho A_2 v C_c$$

where w = air flow through the ring of holes in lb/sec

A_2 = area of the holes in ft^2

ρ = density of air in lb/ft^3

v = velocity of air through holes in ft/sec and is taken as equal to that in the clearance area between the liner and the burner basket

C_o = orifice coefficient

C_o, the orifice coefficient, was obtained from a graph (Ref. 5) after computing Reynold's number and the diameter ratio as follows:

$$A_1 = \frac{\pi}{4}(D_{od}^2 - D_{id}^2)$$

Where A_1 = clearance area between basket and burner shell at the location of the primary holes in ft^2

D_{od} = outside diameter of clearance area in ft

D_{id} = inside diameter of clearance area in ft

$$A_1 = .785\left[\left(\frac{7.25}{12}\right)^2 - \left(\frac{6.1}{12}\right)^2\right]$$

$$= .089 \text{ ft}^2 = 12.8 \text{ in}^2$$

$$D_1 = \sqrt{\frac{A_1}{\pi/4}}$$

Where D_1 = diameter in ft of a pipe with area equivalent to A_1

$$D_1 = \sqrt{\frac{.089}{.785}}$$

$$= \quad .336 \text{ ft} = 4.04 \text{ in}$$

At this point, it is assumed that .25 lb air/sec will have been used previously for cooling and carbon prevention.

$$v_1 = \frac{w_1}{\rho A_1}$$

where v_1 = velocity in ft/sec through clearance area

w_1 = air flow in clearance area in lb/sec

$$v_1 = \frac{3 - .25}{.0965 \times .089} = 320 \text{ ft/sec}$$

By Sutherland's formula:

$$\mu = \mu_r \left(\frac{T_r + K}{T + K}\right)\left(\frac{T}{T_r}\right)^{3/2}$$

where μ = viscosity of the air in poises

μ_r = 170.9×10^{-6} poises

T_r = 273° K

$K = 120$

T = temperature of air = 366.4° K

$$= 170.9 \times 10^{-6} \left(\frac{393}{486.4}\right)\left(\frac{366.4}{273}\right)^{3/2}$$

$$= 2.145 \times 10^{-4} \text{ poises}$$

$$= 2.145 \times 10^{-4} \times 2.09 \times 10^{-3} \text{ slugs/ft-sec}$$

Then,
$$NR = \frac{D_1 v_1 \rho}{\mu g}$$

where NR = Reynold's number

g = gravitational constant = 32.2 ft/sec^2

$$N_R = \frac{.336 \times 320 \times .0965}{.45 \times 10^{-6} \times 32.2}$$

$$= .716 \times 10^{6}$$

$$A_2 = \frac{\pi}{4} N_h D_h^{2}$$

where N_h = number of holes in ring

D_h = diameter of each hole in ft

$$A_2 = .785 \times 8 \times \left(\frac{1/2}{12}\right)^{2}$$

$$= .011 \text{ ft}^2 = 1.56 \text{ in}^2$$

$$\frac{D_2}{D_1} = \sqrt{\frac{A_2}{A_1}} = \sqrt{\frac{1.56}{12.8}} = .35$$

Then, with these values of N_R and D_2/D_1 and the chart of Reference 5, we find C_o = .61.

Thus,

$$W = .0965 \times .011 \times 320 \times .61$$

$$= .2 \text{ lb/sec}$$

$$Q = \frac{W}{\rho}$$

where Q = air flow through ring of holes in ft^3/sec

$$Q = \frac{.2}{.0965} = 2.07 \text{ ft}^3\text{/sec}$$

The friction pressure drop was then calculated by the following formula from Reference 6:

$$\Delta P = \frac{\rho}{2g}\left[\frac{Q}{A_2C_o}\sqrt{1-\left(\frac{A_2}{A_1}\right)^2}\right]^2$$

where ΔP = friction pressure drop in lb/ft^2

$$\Delta P = \frac{.0965}{2 \times 32.2}\left[\frac{2.07}{.011 \times .61}\sqrt{1-\left(\frac{.011}{.089}\right)^2}\right]^2$$

$$= 147 \text{ lb/ft}^2 = 1.02 \text{ lb/in}^2$$

Since this was considered to be a reasonable value for friction pressure drop, it was selected as the value to be used in designing the combustion chamber.

As previously stated, it was assumed that .25 lb air/sec had been used for cooling and carbon prevention. It was decided that the air to cool the dome should be introduced through a ring of eight 3/8-inch holes around the fuel nozzle. The air jets through these holes impinge on slanted vanes which give them a swirling action and direct them onto the dome. The pressure drop equation was used to determine whether the eight 3/8-inch holes would give the required air flow.

$$Q = \frac{A_2C_o}{\sqrt{1-\left(\frac{A_2}{A_1}\right)^2}}\sqrt{2g\left(\frac{\Delta P}{\rho}\right)}$$

where Q = air flow through the eight 3/8-inch holes in ft^3/sec

A_2 = area of the eight 3/8-inch holes in ft^2

A_1 = area of combustion chamber in ft^2

$$A_2 = 8 \times \frac{\pi}{4}\left(\frac{3/8}{12}\right)^2 = .00615 \text{ ft}^2 = .885 \text{ in}^2$$

$$A_1 = \frac{\pi}{4}\,\frac{7.33}{12}^2 = .29\ \text{ft}^2 = 41.8\ \text{in}^2$$

$$Q = \frac{.00615 \times .61}{\sqrt{1 - \left(\frac{.00615}{.29}\right)^2}}\sqrt{64.4\left(\frac{147}{.0765}\right)}$$

$$= 1.17\ \text{ft}^3/\text{sec or } .11\ \text{lb/sec}$$

This value seems reasonable since a portion of the .25 lb air/sec must enter through an annular slot located between the nose hemisphere and the walls of the basket. The desired flow through this slot is then:

$$.25 - .11 = .14\ \text{lb/sec or } 1.45\ \text{ft}^3/\text{sec}$$

The pressure drop equation was used to find the slot area required to obtain this flow. Since the term $\sqrt{1 - (A_2/A_1)^2}$ will be very nearly equal to 1, the pressure drop equation may be used to solve for A_2 thus:

$$A_2 = \frac{Q}{C_o\sqrt{2g\left(\frac{\Delta}{\rho}\right)}}$$

where A_2 = slot area in ft^2

Q = air flow through the slot in ft^3/sec

$$A_2 = \frac{1.45}{.61\sqrt{64.4\left(\frac{147}{.0765}\right)}}$$

$$= .0076\ \text{ft}^2 = 1.09\ \text{in}^2$$

The required thickness of the slot is found as follows:

$$A_2 = \frac{\pi}{4}\left[D^2 - (D - x)^2\right]$$

where D = diameter of burner basket in inches

x = clearance in inches

$$1.09 = \frac{\pi}{4}\left[36 - (6 - x)^2\right]$$

$$x = .116 \text{ in}$$

The slot thickness will then be:

$$\frac{x}{2} = \frac{.116}{2} = .058 \text{ in} \cong 1/16 \text{ in}$$

In order to get the required air flow for the primary section, three rings of eight 1/2-inch holes were required in the burner basket in addition to the ring of eight 3/8-inch holes in the dome and the annular slot.

Design of Secondary Zone

The weight flow through the primary zone is .85 lb/sec. This leaves 2.15 lb/sec to flow through the secondary zone which includes cooling louvres, the tail cone annulus, and the secondary holes.

Louvres

Air introduced through the louvres is intended to cool the basket and to prevent the formation of carbon. The louvres are shaped to deflect the air along the inner surface of the liner, thus keeping the velocity high enough to sweep away any carbon which might form. Three rings of eight louvres each were considered adequate.

Each louvre is 1/2 inch long and 1/16 inch deep. This gives an area of:

$$A = \frac{1}{16} \times \frac{1}{2} = .0312 \text{ in}^2$$

The area of one ring of eight louvres is:

$$A = .0312 \times 8 = .25 \text{ in}^2$$

Now, assuming an orifice coefficient of .61, the flow through the louvres is as follows:

$$Q = A_2 C_0 \sqrt{2g\left(\frac{\Delta P}{\rho}\right)}$$

(The term $1/\sqrt{1 - (A_2/A_1(^2}$ has been omitted since in all cases it is very nearly equal to 1.)

where $Q = \frac{.25}{144} \times .61\sqrt{64.4\left(\frac{147}{.0965}\right)}$

$$= .332 \text{ ft}^3/\text{sec or } .032 \text{ lb/sec}$$

For three rings, the flow will be:

$$3 \times .032 = .096 \text{ lb/sec}$$

Tail Cone Annulus

The flow through the tail cone annulus is necessary to cool the inner liner which is in direct contact with the hot gases. The film of air need not be very large to accomplish this. The 1/16-inch clearance allowed is in accordance with current practice.

According to Reference 7, the pressure drop across the tail cone annulus is given by the following formula:

$$\frac{\Delta P}{\rho} = \frac{f}{4}\ \frac{h}{R}\ \frac{v^2}{2g}$$

where ΔP = friction pressure drop in lb/ft^2 = 147 lb/ft^2

ρ = density of air in lb/ft^3 = .0965 lb/ft^3

f = friction factor

h = length of tail cone in ft = 5/12 ft

R = hydraulic radius in ft

v = average velocity through annular space in ft/sec

Reference 8 gives the relation between f and Reynold's number, NR. NR, in turn, is dependent upon v. Thus, it can be seen that a cut and try calculation is necessary to arrive at values of f and v which will satisfy the above equation. From Reference 7,

$$NR = \frac{4v\rho R}{\mu g}$$

where μ = viscosity in slugs/ft-sec

$$R = \frac{A_{ave}}{WP}$$

Where A_{ave} = average cross-sectional area in ft^2

WP = wetted perimeter in ft

$$A_{ave} = .785\left[\frac{\left[(6.125)^2 - 6^2\right] + \left[(4.125)^2 - 4^2\right]}{2}\right]$$

$$= 1.025\ in^2 = .00711\ ft^2$$

$$WP = \pi\left(D_2 + D_1\right)$$

where D_1 = inside diameter of section at A_{ave} in ft

D_2 = outside diameter of section at A_{ave} in ft

$$D_2 = D_1 + \frac{.125}{12}$$

$$.00711 = .785\left[\left(D_1 + \frac{.125}{12}\right)^2 - {D_1}^2\right]$$

$$D_1 = .43 \text{ ft}$$

$$D_2 = .44 \text{ ft}$$

$$WP = \pi(.43 + .44) = 2.735 \text{ ft}$$

$$R = \frac{.00711}{2.735} = .0026 \text{ ft}$$

Assuming v = 250 ft/sec:

$$NR = \frac{4 \times 250 \times .0965 \times .0026}{.45 \times 10^{-6} \times 32.2}$$

$$= 17320$$

and f = .042

Solving for v to check assumption:

$$\frac{147}{.0965} = \frac{.042}{4}\left(\frac{3/12}{.0026}\right)\frac{v^2}{64.4}$$

$$v = 242 \text{ ft/sec}$$

Assuming v = 240 ft/sec and recalculating:

$$NR = 17320 \times \frac{240}{250} = 13860$$

and f = .0435

$$v = 242\sqrt{\frac{.042}{.0435}} = 238 \text{ ft/sec}$$

This checks the assumed value. Then,

$$Q = A_{ave}\ v$$

where Q = flow through tail cone annulus in ft^3/sec

$$Q = .00711 \times 240$$

$$= 1.707 \text{ ft}^3\text{/sec or } .165 \text{ lb/sec}$$

Secondary Holes

Knowing the flow through the primary zone, the louvres, and the tail cone annulus, the weight flow through the secondary holes can be obtained.

.85 lb/sec	primary air
.165 lb/sec	tail cone annulus
.096 lb/sec	louvres
1.111 lb/sec	total

Then, the weight flow through the secondary holes is

$$3.000 - 1.111 = 1.889 \text{ lb/sec}$$

Because of the high velocities in the combustion chamber, it was assumed that the air would flow through the secondary holes at some angle other than 90°. This angle was taken to be 45° throughout the secondary zone. Appendix A presents an experiment which was the basis for this assumption.

This angularity of flow necessitates the introduction of an effectiveness coefficient, C_e, in addition to the orifice coefficient, C_o, when calculating flow through

a secondary hole. C_e is defined as the sine of the angle of flow.

$$C_e = \sin 45^\circ$$

$$= .707$$

To find the flow through a ring of eight 5/8-inch secondary holes:

$$Q = N_h A_2 C_e C_o \sqrt{2g\left(\frac{\Delta P}{\rho}\right)}$$

where Q = air flow through ring of holes in ft^3/sec

N_h = number of holes in ring = 8

A_2 = area of each hole = .00213 ft^2

C_e = effectiveness coefficient = .707

C_o = orifice coefficient = .61

g = gravitational constant = 32.2 ft/sec^2

ΔP = friction pressure drop = 147 lb/ft^2

ρ = density of air = .0965 lb/ft^3

$$Q = 8 \times .00213 \times .707 \times .61 \sqrt{64.4\left(\frac{147}{.0965}\right)}$$

$$= 2.29 \text{ ft}^3/\text{sec or } .221 \text{ lb/sec}$$

Similarly, Q for a ring of eight 11/16-inch holes is 2.88 ft^3/sec or .278 lb/sec. Thus, six rings of 11/16-inch holes and one ring of 5/8-inch holes will admit the 1.889 lb/sec of secondary air that remains.

Check of Air Flows

	.11	lb/sec	holes in hemisphere
	.14	"	annular slot
3 x .2 =	.60	"	three rings of 1/2-inch holes
6 x .278 =	1.67	"	six rings of 11/16-inch holes
	.221	"	one ring of 3/8-inch holes
	.165	"	tail cone annulus
	.096	"	louvres
	3.002	lb/sec	total air flow check

Location of Holes

The primary air should be introduced far enough downstream to insure the development of a strong combustion eddy. This is about three-fourths to one diameter from the fuel nozzle.

The secondary air should be first introduced at a low rate and then at an increasingly greater rate downstream. This is to prevent chilling of the flame front and the resulting stoppage of the combustion process. By introducing the first part of the secondary air six to eight inches from the fuel nozzle, enough time is allowed for the combustion to be well along towards completion.

A convenient method for locating the primary and secondary holes to satisfy these requirements is shown in Figure 6 in which total area of the openings into the basket

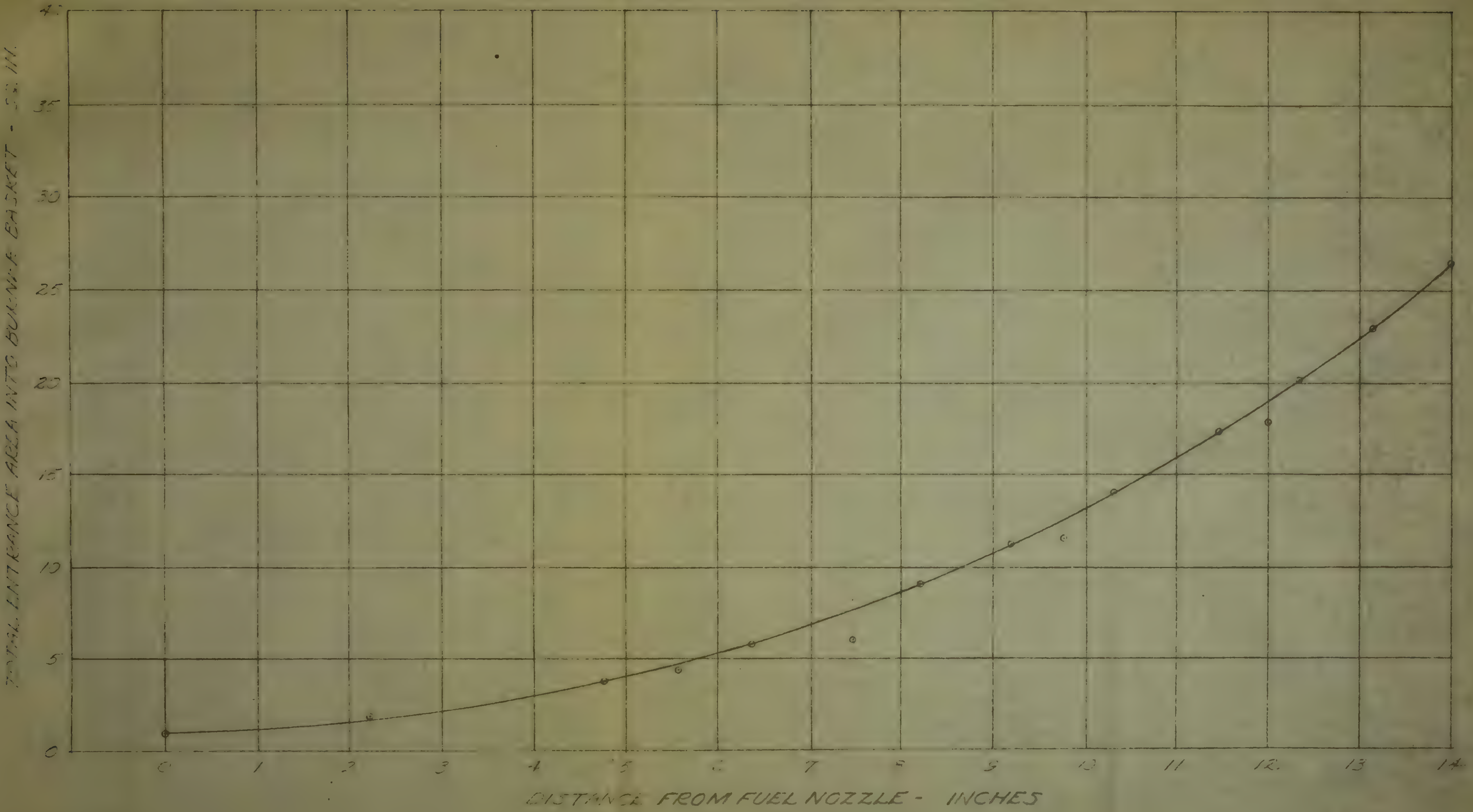

FIGURE 6
CURVE USED IN LOCATING PRIMARY AND SECONDARY HOLES.

is plotted vs. distance from the fuel nozzle. The curve shown in an arc of a circle and is defined as follows:

1. The center of the circle lies on the vertical line through the zero distance point.
2. One point on the arc is the area of the openings into the burner basket ahead of the fuel nozzle, i.e., the area of the eight 3/8-inch holes. This area is plotted at the point of zero distance.
3. The other point of the arc is the total area of all the openings, slots, and louvres in the burner basket. This area is plotted at the distance at which the last row of secondary holes is desired.

Now, by taking the areas of the rings of holes and fitting them to this curve, a satisfactory distribution will automatically follow.

This method, used judiciously, should take much of the cut and try out of burner design.

The cooling louvres can now be placed in spots where the boundary layer would otherwise thicken up and let carbon deposits form.

Figure 7 shows the final configuration of the combustion chamber.

FIGURE 7 DESIGNED COMBUSTION CHAMBER.

FULL SCALE

VII. CALCULATION OF PERFORMANCE

Friction Pressure Drop

Since the flow through the combustion chamber is parallel, the pressure drop through any one part is equal to the pressure drop through the entire burner. The friction pressure drop, as calculated in the preceding section, is:

$$\Delta P = 147 \text{ lb/ft}^2 \quad \text{or} \quad 1.02 \text{ lb/in}^2$$

Momentum Pressure Drop

Having the friction pressure loss in hand, it is now necessary to determine the pressure loss due to combustion in order to have the total pressure loss through the burner.

The assumption is now made that the combustion takes place in a burner of constant cross-sectional area and that no mass is added by the fuel.

In a combustion process, the impulse-momentum law holds. This law states that "impulse is equal to the change in momentum." Written mathematically, this is:

$$Ft = \Delta (Mv)$$

where F = pounds force

t = time in sec

M = pounds mass

v = velocity in ft/sec

Now, since mass is assumed constant:

$$Ft = M\Delta v$$

$$F = \frac{M\Delta v}{t}$$

$$F = \frac{w}{g}(v_5 - v_4)$$

where w = weight flow in lb/sec

g = ratio of absolute to gravitational units of mass

Pressure is force per unit area, and since the burner is of constant area,

$$AP_{4s} - AP_{5s} = \frac{w}{g}(v_5 - v_4)$$

where P_{4s} = entering static pressure in lb/ft^2

P_{5s} = exit static pressure in lb/ft^2

A = burner area in ft^2

In the burner at hand, the known conditions are: entering total pressure, P_{4t}; entering velocity, v_4; entering total temperature, T_{4t}; and exit total temperature, T_{5t}. In order to obtain pressure drop due to combustion, it is necessary to know the theoretical exit total pressure, P_{5t}.

From adiabatic considerations, it is known that:

$$\frac{P_{4t}}{P_{4s}} = \left(\frac{T_{4t}}{T_{4s}}\right)^{\frac{\gamma}{\gamma-1}}$$

and $$\frac{P_{5t}}{P_{5s}} = \left(\frac{T_{5t}}{T_{5s}}\right)^{\frac{\gamma}{\gamma-1}}$$

Then, $$\frac{P_{5t}}{P_{4t}} = \frac{P_{5s}}{P_{4s}}\left(\frac{T_{5t}}{T_{4t}}\right)^{\frac{\gamma}{\gamma-1}}\left(\frac{T_{4s}}{T_{5s}}\right)^{\frac{\gamma}{\gamma-1}}$$

Since $$P/\rho = RT$$

where P = pressure in lb/ft^2

ρ = density in lb/ft^3

R = gas constant in ft/°R

T = temperature in °R

and $$Q = w/\rho$$

where Q = volume flow in ft^3/sec

then $$\frac{PQ}{w} = RT$$

and $$\frac{P_{4s}Q_4}{T_{4s}} = Rw = \frac{P_{5s}Q_5}{T_{5s}}$$

Also, $$\frac{P_{4s}V_4}{T_{4s}} = \frac{P_{5s}V_5}{T_{5s}}$$

So, $$\frac{P_{5t}}{P_{4t}} = \frac{V_4}{V_5}\left(\frac{T_{5t}}{T_{4t}}\right)^{\frac{\gamma}{\gamma-1}}\left(\frac{T_{4s}}{T_{5s}}\right)^{\frac{\gamma}{\gamma-1}}$$

To find static temperatures:

$$\frac{T_{4t}}{T_{4s}} = 1 + \frac{\gamma - 1}{2} M_4^2$$

and $$M_4^2 = \frac{V_4^2}{\gamma g R T_{4s}}$$

$$\frac{T_{4t}}{T_{4s}} = 1 + \frac{\gamma - 1}{2}\left(\frac{v_4^2}{\gamma g R T_{4s}}\right)$$

$$T_{4t} = T_{4s} + \frac{\gamma - 1}{2}\left(\frac{v_4^2}{\gamma g R}\right)$$

Then, $T_{4s} = T_{4t} - \dfrac{v_4^2}{2gR\left(\dfrac{\gamma}{\gamma - 1}\right)}$

Similarly, T_{5s} is obtained.

To find v_5:

$$AP_{4s} - AP_{5s} = \frac{w}{g}(v_5 - v_4)$$

$$A = Q/v$$

$$\frac{P_{4s}Q_4}{v_4} - \frac{P_{5s}Q_5}{v_5} = \frac{w}{g}(v_5 - v_4)$$

$$P/\rho = RT$$

$$Q = w/\rho$$

$$PQ = RTw$$

Now, $\dfrac{RT_{4s}w}{v_4} - \dfrac{RT_{5s}w}{v_5} = \dfrac{w}{g}(v_5 - v_4)$

$$\frac{T_{4s}}{v_4} - \frac{T_{5s}}{v_5} = \frac{1}{Rg}(v_5 - v_4)$$

$$\frac{T_{4s}}{v_4} = \frac{T_{4t}}{v_4} - \frac{v_4}{2gR\left(\frac{\gamma}{\gamma - 1}\right)}$$

$$\frac{T_{5s}}{v_5} = \frac{T_{5t}}{v_5} - \frac{v_5}{2gR\left(\frac{\gamma}{\gamma - 1}\right)}$$

$$\frac{T_{4s}}{v_4} - \frac{T_{5s}}{v_5} = \frac{T_{4t}}{v_4} - \frac{T_{5t}}{v_5} + \frac{v_5 - v_4}{2gR\left(\frac{\gamma}{\gamma - 1}\right)}$$

$$\frac{T_{4t}}{v_4} - \frac{T_{5t}}{v_5} + \frac{v_5 - v_4}{2gR\left(\frac{\gamma}{\gamma - 1}\right)} = \frac{1}{Rg}(v_5 - v_4)$$

$$\frac{T_{4t}}{v_4} - \frac{T_{5t}}{v_5} = \frac{1}{Rg}(v_5 - v_4) - \frac{v_5 - v_4}{2gR\left(\frac{\gamma}{\gamma - 1}\right)}$$

$$= (v_5 - v_4)\left[\frac{1}{Rg} - \frac{1}{2gR\left(\frac{\gamma}{\gamma - 1}\right)}\right]$$

$$= (v_5 - v_4)\left[\frac{2\gamma - (\gamma - 1)}{2\gamma gR}\right]$$

$$= (v_5 - v_4)\left(\frac{\gamma + 1}{2\gamma gR}\right)$$

$$\frac{T_{4t}}{v_4}\left[1 - \frac{T_{5t}/v_5}{T_{4t}/v_4}\right] = \left(\frac{\gamma + 1}{2\gamma gR}\right) v_4 \left(\frac{v_5}{v_4} - 1\right)$$

$$\frac{T_{4t}}{v_4}\left[1 - \frac{T_{5t}}{T_{4t}}\frac{v_4}{v_5}\right] = \left(\frac{\gamma + 1}{2\gamma gR}\right) v_4 \left(\frac{v_5}{v_4} - 1\right)$$

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$$1 - \frac{T_{5t}}{T_{4t}}\frac{V_4}{V_5} = \left(\frac{\gamma + 1}{2}\right)\left(\frac{V_4^{\,2}}{\gamma gRT_{4t}}\right)\left(\frac{V_5}{V_4} - 1\right)$$

$$= \left(\frac{\gamma + 1}{2}\right) M_{4t}^{2}\left(\frac{V_5}{V_4} - 1\right)$$

$$= \left(\frac{\gamma + 1}{2}\right) M_{4t}^{2}\left(\frac{V_5}{V_4}\right) - \left(\frac{\gamma + 1}{2}\right) M_{4t}^{2}$$

$$1 + \left(\frac{\gamma + 1}{2}\right) M_{4t}^{2} = \left(\frac{\gamma + 1}{2}\right) M_{4t}^{2}\left(\frac{V_5}{V_4}\right) + \frac{T_{5t}/T_{4t}}{V_5/V_4}$$

$$(V_5/V_4)^2\left(\frac{\gamma + 1}{2}\right) M_{4t}^{2} - \frac{V_5}{V_4}\left[1 + \left(\frac{\gamma + 1}{2}\right) M_{4t}^{2}\right] + \frac{T_{5t}}{T_{4t}} = 0$$

let $a = \frac{\gamma + 1}{2} M_{4t}^{2}$ and $b = \frac{T_{5t}}{T_{4t}}$

$$a\left(\frac{V_5}{V_4}\right)^2 - (1 + a)\frac{V_5}{V_4} + b = 0$$

$$\frac{V_5}{V_4} = \frac{1 + a \pm \sqrt{(1 + a)^2 - 4ab}}{2a}$$

then,

$$\frac{V_5}{V_4} = \frac{1 + \frac{\gamma + 1}{2} M_{4t}^{2} \pm \sqrt{1 + \frac{\gamma + 1}{2} M_{4t}^{2}\,{}^{2} - 4\left(\frac{\gamma + 1}{2} M_{4t}^{2}\right)\frac{T_{5t}}{T_{4t}}}}{(\gamma + 1) M_{4t}^{2}}$$

Substituting the following values in the formula derived for v_5:

$v_4 = 100$ ft/sec　　$M_{4t} = .0794$

$\gamma = 1.4$　　$M_{4t}^2 = .0063$

$\gamma + 1 = 2.4$　　$\frac{T_{5t}}{T_{4t}} = \frac{1960}{660} = 2.97$

$\frac{\gamma + 1}{2} = 1.2$

we find,

$$\frac{v_5}{100} = \frac{1 + 1.2(.0063) \pm \sqrt{\left[1 + 1.2(.0063)\right]^2 - 4(1.2)(.0063)2.97}}{2.4(.0063}$$

$$v_5 = 302 \text{ ft/sec}$$

Substituting the following values in the formulas derived for T_{4s} and T_{5s}:

$v_4 = 100$ ft/sec　　$v_5 = 302$ ft/sec

$g = 32.2$ ft/sec^2　　$\gamma = 1.4$

$T_{4t} = 660°$ R　　$T_{5t} = 1960°$ R

$R = 53.3$ ft/°R

we find,

$$T_{4s} = 660 - \frac{(100)^2}{2(32.2)\left(\frac{1.4}{1.4 - 1}\right)53.3}$$

$$= 659.2° \text{ R}$$

and

$$T_{5s} = 1960 - \frac{(302)^2}{2(32.2)\left(\frac{1.4}{1.4 - 1}\right)53.3}$$

$$= 1952.4° \text{ R}$$

Substituting the following values in the formula derived for P_{5t}:

P_{4t} = 1.6 atm $\qquad \gamma$ = 1.4

v_4 = 100 ft/sec $\qquad v_5$ = 302 ft/sec

T_{4t} = 660° R $\qquad T_{5t}$ = 1960° R

T_{4s} = 659.2° R $\qquad T_{5s}$ = 1952.4° R

we find,

$$\frac{P_{5t}}{1.6} = \frac{\left(\frac{100}{302}\right)\left(\frac{1960}{660}\right)^{\frac{1.4}{1.4-1}}}{\left(\frac{1952.4}{659.2}\right)^{\frac{1}{1.4-1}}}$$

P_{5t} = 1.52 atm

Then, the momentum pressure drop is:

$$\Delta P = P_{4t} - P_{5t}$$

$$= 1.6 - 1.52$$

$$= .08 \text{ atm} = 1.175 \text{ lb/in}^2$$

Total Pressure Drop

The total pressure drop is the sum of the friction pressure drop and the momentum pressure drop.

Friction pressure drop = 1.02 lb/in^2

Momentum pressure drop = 1.175 lb/in^2

Total pressure drop = 2.195 lb/in^2

For ease in comparing burners of various sizes and capacities, the parameter $\Delta P/P_{4t}$ is used. This value usually lies between four per cent and seven per cent.

$$\frac{\Delta P}{P_{4t}} = \frac{2.195}{1.6 \times 14.7} = .0934 \text{ or } 9.34\%$$

This value is a little high but is explained by the very low value of P_{4t}.

Reference 3 states that the total pressure drop should be in the neighborhood of 20 to 30 times the value of the velocity head q at point 4.

$$q = \frac{\rho v^2}{2g}$$

$$= \frac{.0965(100)^2}{64.4}$$

$$= 15 \text{ lb/ft}^2 \text{ or } .104 \text{ lb/in}^2$$

$$\frac{\Delta P}{q} = \frac{2.195}{.104} = 21.1$$

Thus it is seen that the pressure drop obtained is in accord with Reference 3.

Combustion Efficiency

In an unpublished paper, Mr. W. R. Hawthorne, in collaboration with Professor H. C. Hottel of the Massachusetts Institute of Technology, presented an empir-

ical formula for combustion efficiency. This formula was derived from curves of a series of combustion chambers showing the correlation between combustion efficiency, heat input, and diameter of chamber. The formula is as follows:

$$1 - \eta = e^{\frac{-6 \times 10^6}{ID}}$$

where η = combustion efficiency

e = natural log base = 2.718

I = combustion intensity in Btu/hr-ft^3-atm

D = diameter of combustion chamber in ft

$$1 - \eta = (2.718)^{\frac{-6 \times 10^6}{8 \times 10^6 \times .61}}$$

$$= (2.718)^{-1.235}$$

$$= \frac{1}{3.44} = .2905$$

$$\eta = 1 - .2905$$

$$= .7095 \text{ or } 70.95\%$$

This is a low combustion efficiency. However, this can be attributed to the fact that the very low value of P_{4t} necessitates a very high combustion intensity. The effect of combustion intensity can be shown by substituting an intensity of 4 x 10^6 Btu/hr-ft^3-atm in the empirical formula. In this case, the efficiency jumps to 92%. In order to obtain a combustion intensity of 4 x 10^6, P_{4t} must equal 3.2 atmospheres. Even this value of P_{4t} is considered quite low for modern turbo-jet engines.

VIII. CONCLUSIONS

The combustion chamber resulting from this design shows a total pressure drop of 2.195 lb/in^2. G. Geoffrey Smith in Reference 9 states that several standard turbojet engines have combustion chambers which operate with a total pressure drop of about 2 lb/in^2. From this, it would seem that the pressure drop for the designed combustion chamber was not excessive.

The British use pressure drop divided by reference velocity head ($\Delta P/q$) as a parameter for comparing different combustion chambers. Values of this parameter from 20 to 30 are considered reasonable. Here again, the designed combustion chamber conforms, having a $\Delta P/q$ of 21.1.

Another parameter is pressure drop divided by total inlet pressure ($\Delta P/P_{4t}$). Most values of this parameter quoted for standard combustion chambers varied from three to six per cent. However, these values were invariably for total inlet pressures of 3.5 to 4.5 atmospheres. A test of a DeHavilland H-1 combustion chamber, operating at an inlet pressure of 1.8 atmospheres (a value very near the design point of 1.6 atmospheres), showed a $\Delta P/P_{4t}$ of 18%. Thus, it would appear that the value of 9.34% calculated for the designed chamber is acceptable.

Hawthorne's empirical formula for combustion efficiency was given with no indication as to its valid range.

it may be that using this formula under a condition of low inlet pressure is not justified and that actual tests of the chamber would prove the value of 70.95% obtained from it to be much too low a value of combustion efficiency. This would appear to be the case, judging by the results obtained in the aforementioned test of the DeHavilland chamber. Operating under conditions very close to the design point used in this investigation, an efficiency of 94% was obtained.

It is regrettable that time did not permit the building and testing of the design. Only with the test results in hand can it be said positively that the design is good or bad. However, by comparison with proved designs, it would appear to be a satisfactory combustion chamber.

APPENDIX A

OBSERVATION OF FLOW THROUGH SECONDARY HOLES OF TURBO-JET COMBUSTION CHAMBER

Summary

In order to observe the angularity of flow through the secondary holes of a burner basket, a test box was constructed with entrance, exit, and hole areas similar to those in the combustion chamber of a J-33 turbo-jet engine. The J-33 chamber was selected since it was found to be the most nearly similar to the burner designed in this investigation.

Tufts were suspended from the centers of the holes, and the angle of flow was taken as the angle measured between the tufts and the plate containing the holes during passage of an air stream through the test box.

An attempt to read the flow angle by introducing smoke into the air stream was a failure due to the high velocity and the large amount of turbulence present.

It was concluded that, while the angle of flow varied with distance from the fuel nozzle, an average angle of 45° could be used in determining the area of the secondary holes.

Introduction

If the flow through a hole is at an angle other than 90°, the area allowing fluid to pass will not be the ob-

served area of the hole but the projection of the area on the plane perpendicular to the flow. The formula for flow through an orifice is derived for a flow perpendicular to the orifice. When the flow is not perpendicular to the orifice, the projected area must be used in order for the orifice formula to be valid.

The projected area is

$$A = A_h \sin \theta$$

where A = projected area

A_h = area of hole

θ = angle of flow

In this paper, $\sin \theta$ is called C_e, effectiveness coefficient.

Equipment

Figures 8 and 9 show general views of the equipment used in the experiment. In these figures, the parts are numbered as follows:

(1) Eight-inch pipe carrying the flow from the rotary compressor located in the basement.

(2) Water manometer connected to pitot-static tube in the pipe.

(3) Compressed air line.

(4) Smoke generator.

(5) Test box.

Figure 10 shows a close-up of the smoke generator.

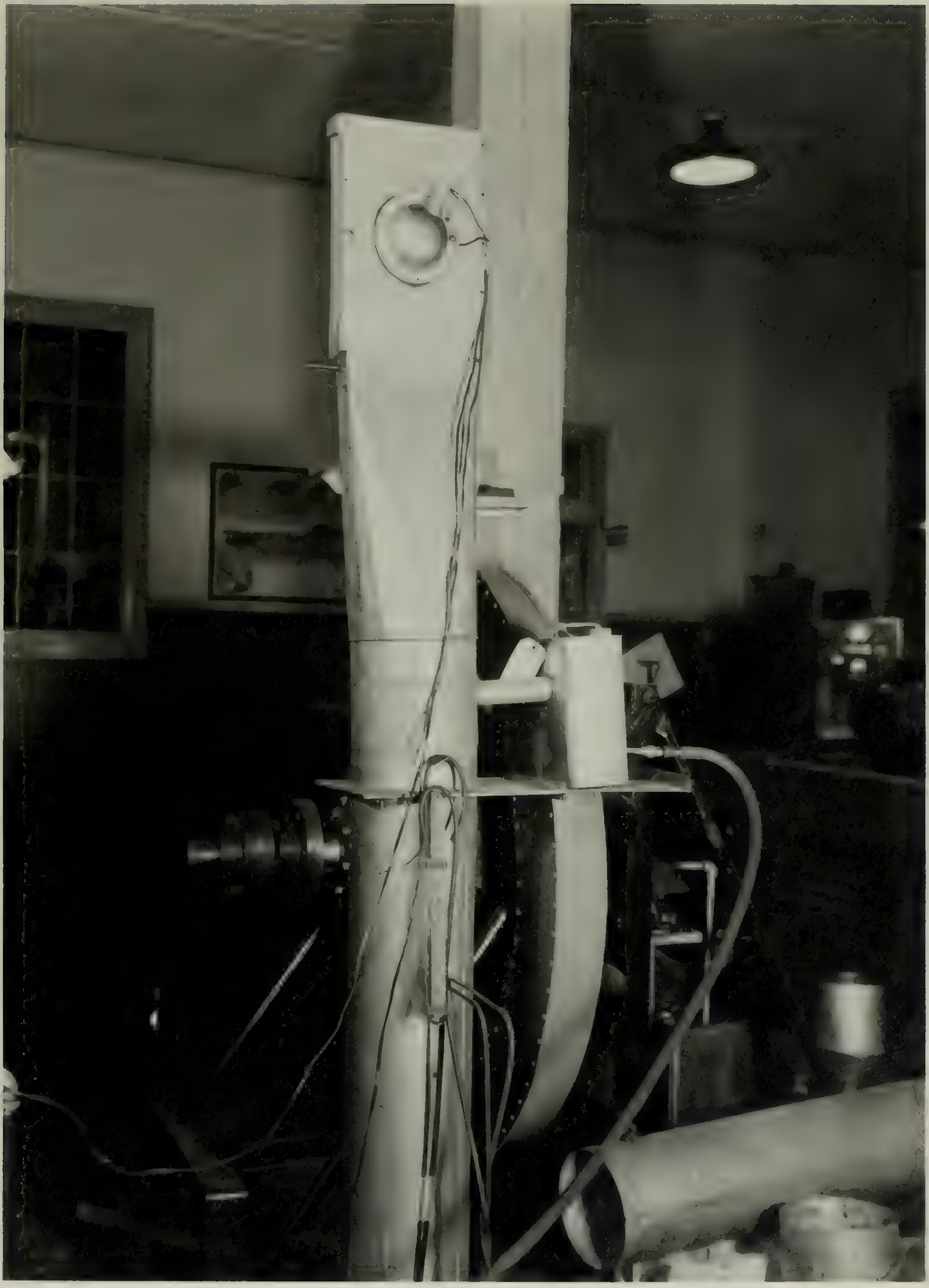

Figure 8. Photograph of Test Setup for Observing Angle of Flow.

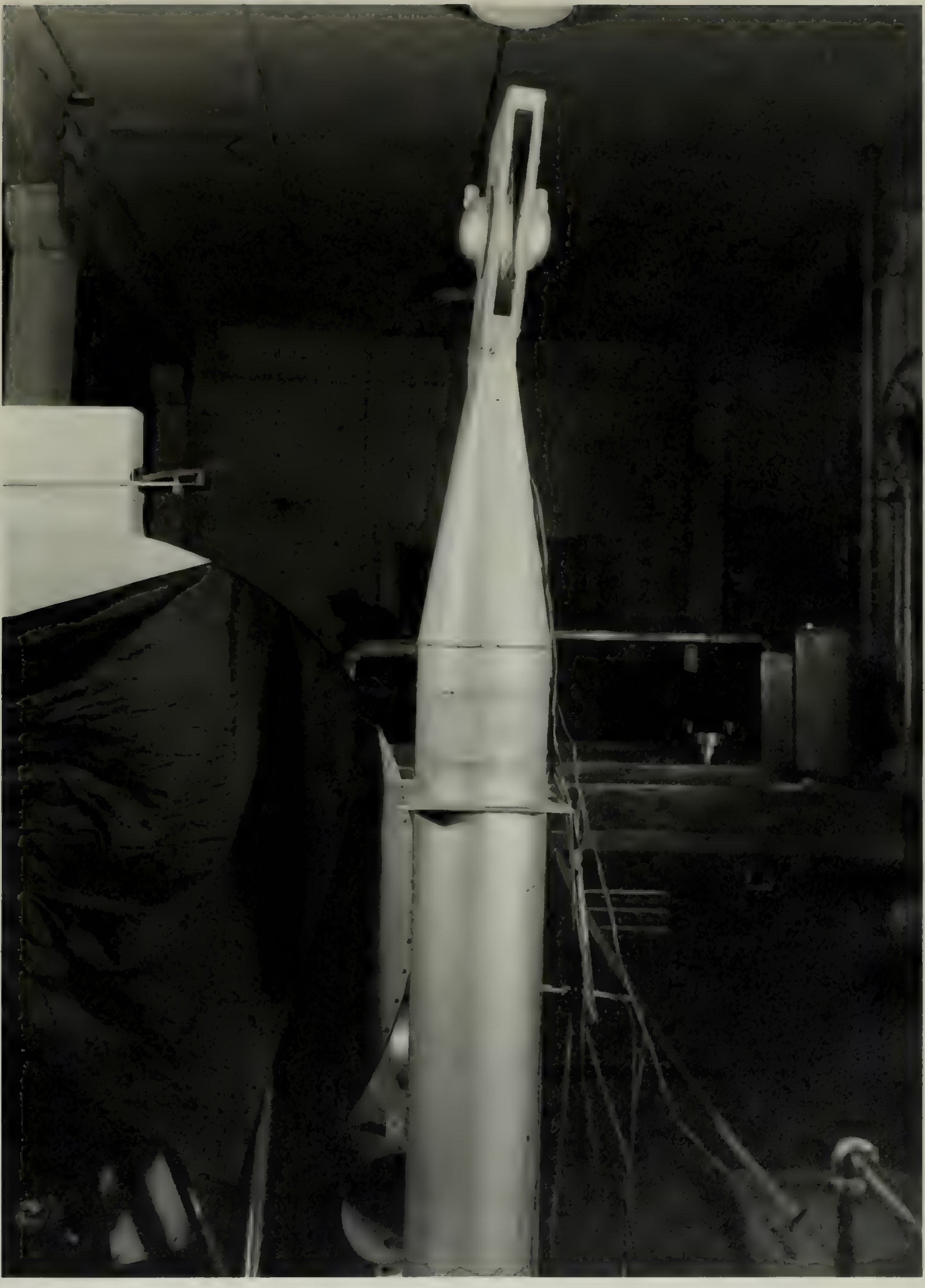

Figure 9. Another View of the Test Setup for Observing Angle of Flow.

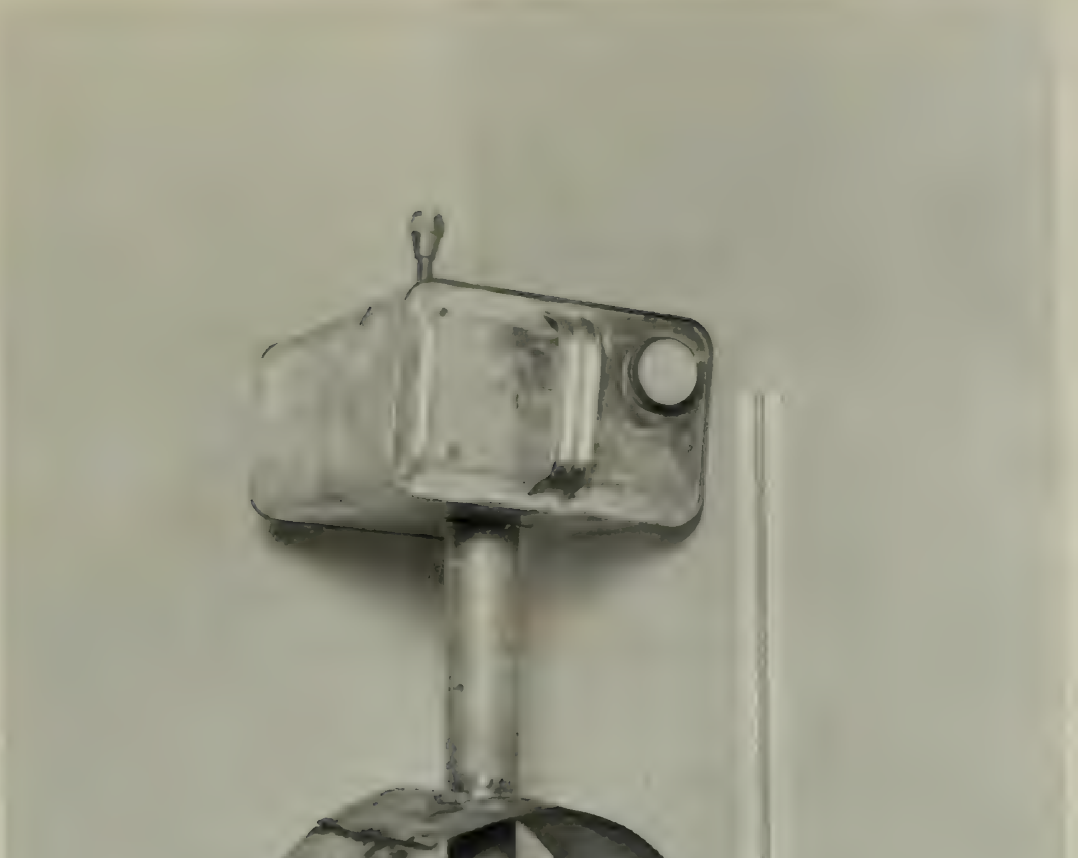

Figure 10. Close-up View of Smoke Generator.

Titanuim tetra-chloride ($TiCl_4$) was placed in the gallon can to produce smoke. The amount of smoke was regulated by a valve in the high pressure air line.

Figure 11, a close-up of the test box, shows the window and the lights used in taking pictures of the flow. The test box itself was designed to simulate conditions in a J-33 burner. Its length was the same as the length of the J-33 burner basket. The plate with the holes in it was placed in such a way that it divided the inlet of the box into two areas: one area equal to the clearance area between the outer shell and the burner basket and the other area equal to the area of the holes around the fuel nozzle plus the area of the annular space between the nose hemisphere and the burner basket. In this manner, approximately correct proportions of the flow were introduced on each side of the plate. At the exit to the test box, the plate again divided the flow. The small area at the exit was made equal to the area between the inner and outer walls of the tail cone. The plate was made with the same size and number of holes as the burner basket.

Fine wire was glued across representative holes in the plate and silk thread tied loosely to it. These silk tufts alined themselves with the flow, making it possible to measure the flow angle.

Procedure

The compressor was started, and, when the flow had

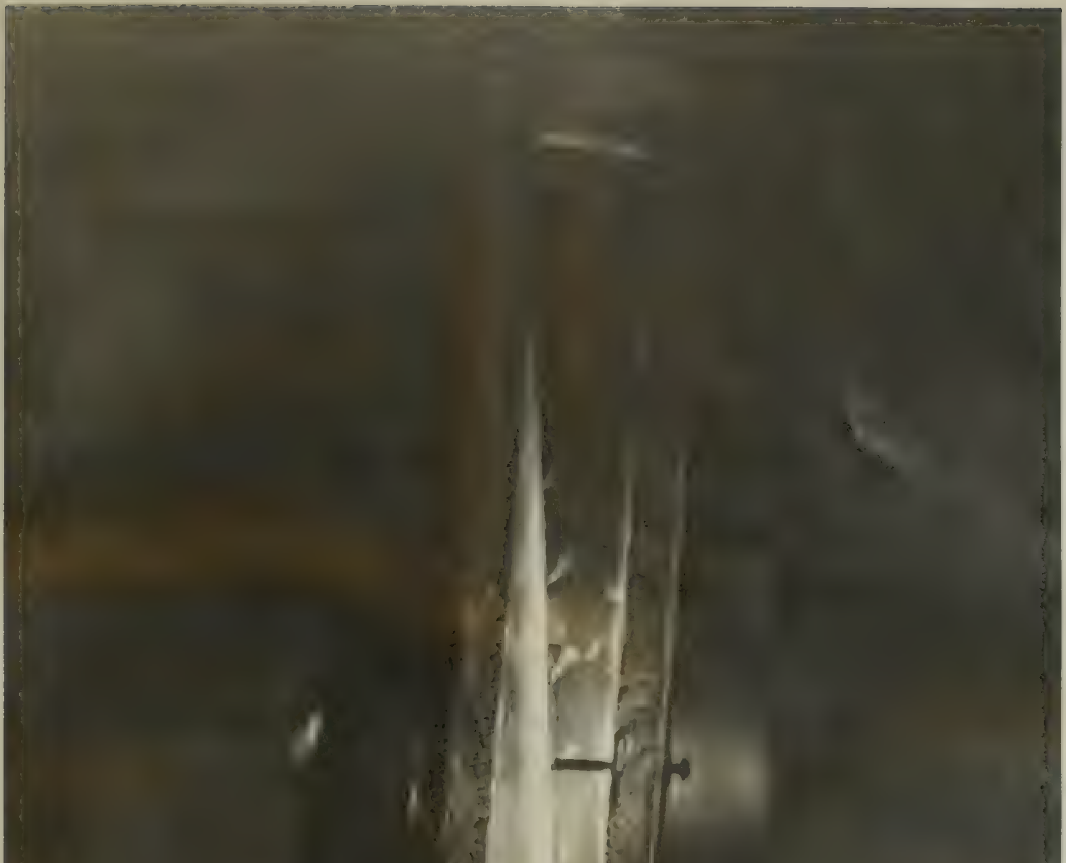

Figure 11. Close-up View of Test Box.

steadied, readings were taken of the temperature and of the pressure measured by the pitot-static manometer. Then the angle of the tufts was recorded. This was done for three different flows. Photographic records were taken of all runs.

The smoke generator was operated as follows: About 5 cc of $TiCl_4$ were put in the gallon can before the run was started. When smoke was desired, the valve in the high pressure air line was opened. Varying the amount of high pressure air very effectively regulated the amount of smoke obtained. This arrangement gave a very satisfactory supply of smoke under good control. Unfortunately, it was impossible to detect the angle of flow through the holes by this means due to the high velocities and large amount of turbulence.

Results

Table II and Figure 12 show the results of the tests. Photographs taken during the runs are shown in Figures 13, 14, and 15. From the results it can be seen that the flow angle varies from 60° to 35° as the distance downstream increases. There is little variation with change in velocity. The average flow angle over the length of the secondary zone was approximately 48°.

Conclusions

Since the average flow angle is 45°, it is apparent that an effectiveness coefficient is necessary for calculating the flow through the secondary holes. In this analytical design, it seemed reasonable to use the average value of flow angle of 45° over the entire length of the secondary zone. Thus, the effectiveness coefficient is

$$C_e = \sin 45^\circ$$

$$= .707$$

Table 11. Flow Angle Through Holes for Various Weight Flows.

Hole No.*	Flow Angle - Degrees		
	Weight Flow - lb/sec		
	1.18	1.32	1.46
1	51	62	60
2	45	52	49
3	45	46	46
4	45	45	45
5	42	42	42
6	40	40	40
7	40	37	38
8	38	35	37
9	37	38	38

* Hole number 1 is the first 5/8-inch hole in the test plate. It is located four inches from the entrance to the test box. The successive holes are one inch apart.

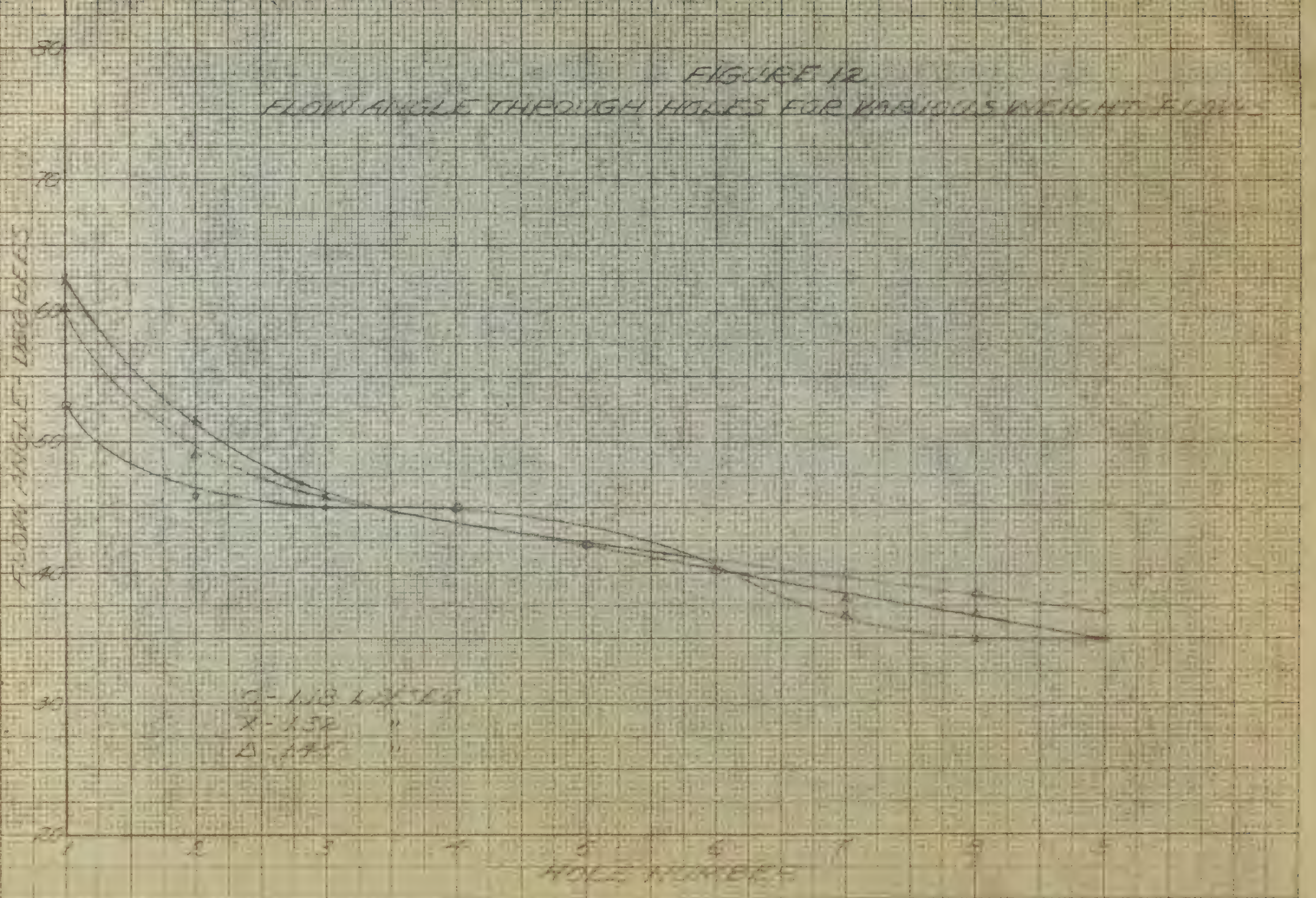

FIGURE 12

FLOW ANGLE THROUGH HOLES FOR VARIOUS WEIGHT FLOW

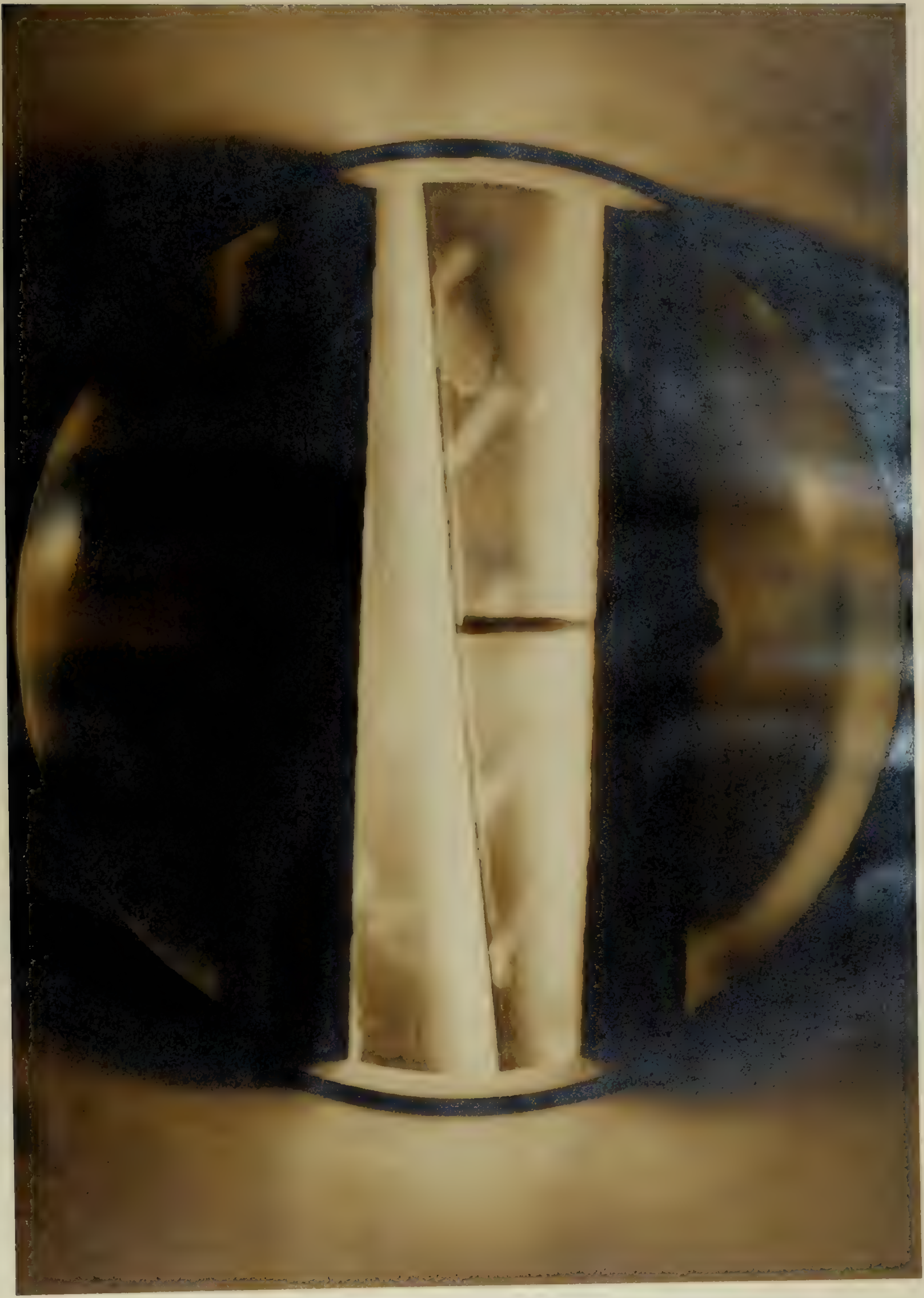

Figure 13. Interior of Test Box Showing Angle of Silk Tufts During Run at 1.18 lb/Sec.

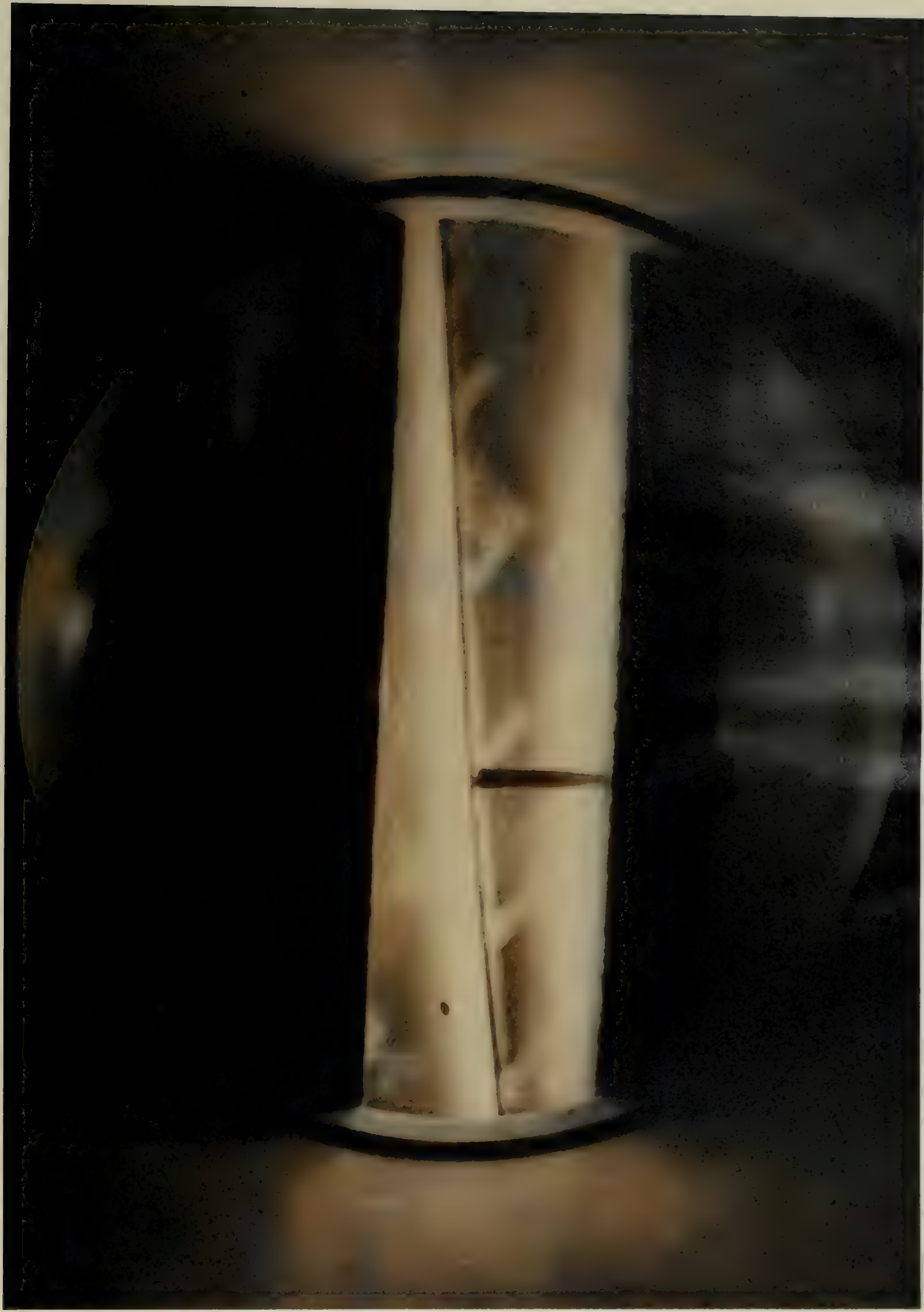

Figure 14. Interior of Test Box Showing Angle of Silk Tufts During Run at 1.45 Lb/Sec.

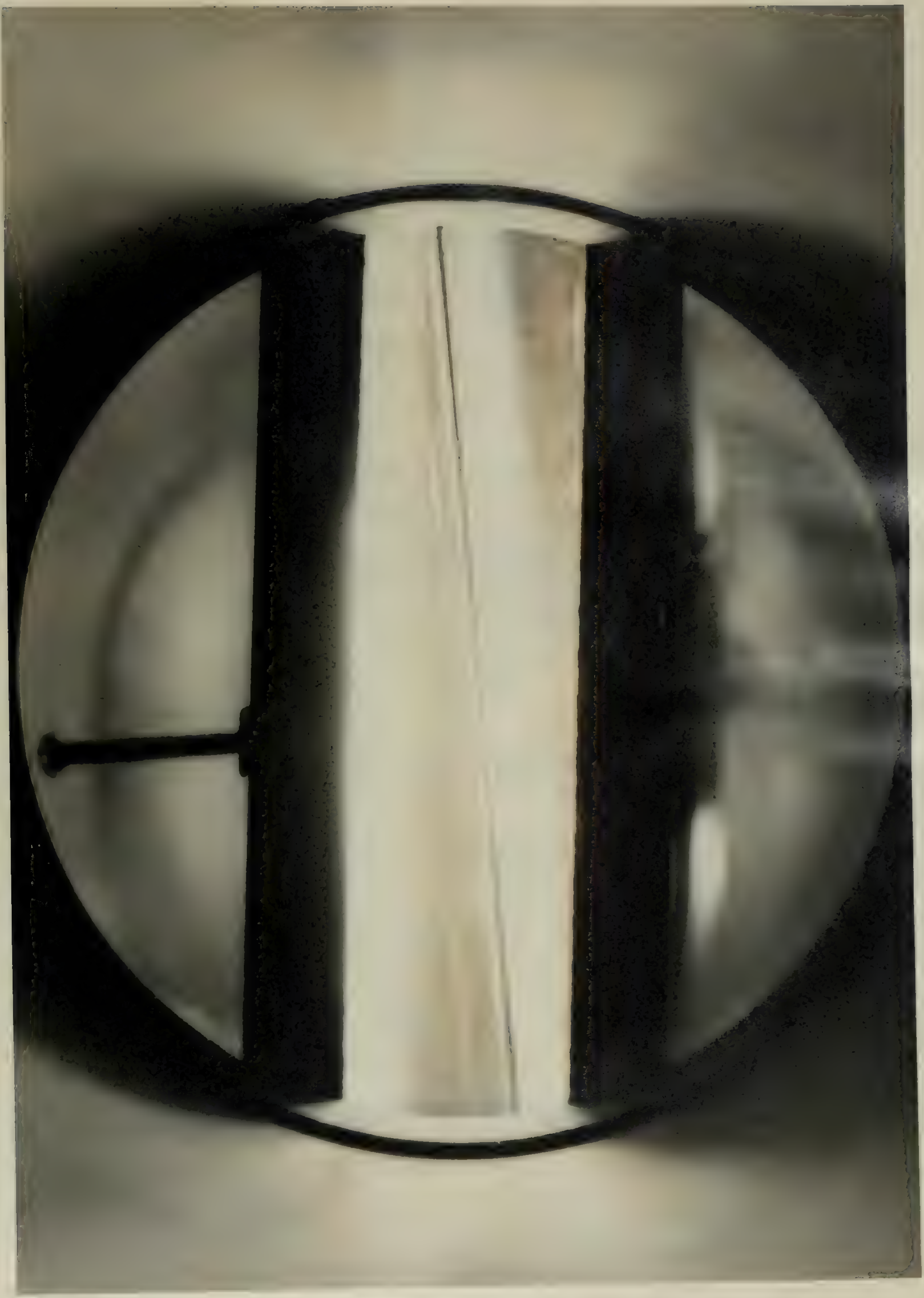

Figure 15. Interior of Test Box During Run in Which Smoke was Used.

Sample Calculations

To find the weight flow through the test box when conditions in the pipe are:

Δh = pitot-static manometer reading = 1/2" H_2O = 2.6 lb/ft^2

Temperature = 130° F or 590° R

$$w = \rho A V$$

where w = weight flow in lb/sec

ρ = density in lb/ft^3

A = area of the eight-inch pipe in ft^2

V = velocity in ft/sec

$$\rho = .0765 \times \frac{520}{590} = .0675 \text{ lb/ft}^3$$

$$A = .785 \times \frac{8^2}{144} = .35 \text{ ft}^2$$

$$V = \sqrt{\frac{2g \times \Delta h}{\rho}}$$

$$= \sqrt{\frac{64.4 \times 2.6}{.0675}}$$

$$= 49.9 \text{ ft/sec}$$

Then, $w = .0675 \times .35 \times 49.9$

$$= 1.18 \text{ lb/sec}$$

APPENDIX B

SYMBOLS

A - maximum area of combustion chamber

A_1 - clearance area between basket and outer liner

A_2 - entrance area into basket

a - speed of sound

C_e - effectiveness coefficient

C_o - orifice coefficient

$c_{p\,avg}$ - average specific heat at constant pressure

c_{p4} - specific heat at constant pressure at inlet

c_{p5} - specific heat at constant pressure at exit

D - diameter of basket

D_1 - diameter of pipe with area equivalent to A_1

D_h - diameter of each hole in ring

D_{max} - maximum diameter of combustion chamber

D_{id} - inside diameter of clearance area

D_{od} - outside diameter of clearance area

d - minor diameter of tail cone

f - friction factor

g - gravitational constant

h - length of tail cone

I - combustion intensity

L - length from nozzle to turbine entrance

l - length from nozzle to end of basket

l_c - length of cylinder or basket

LHV - lower heating value of fuel

M_r - reference Mach number

M_4 - entrance Mach number

M_5 - exit Mach number

N - number of combustion chambers

N_h - number of holes in ring around basket

NR - Reynolds number

P - pressure

P_{sc} - atmospheric pressure at standard conditions

P_{4s} - entering static pressure

P_{5s} - exit static pressure

ΔP - pressure drop

Q - volume air flow

q - heat input

R - gas constant

T_{sc} - temperature at standard conditions

T_{4s} - entrance static temperature

T_{5s} - exit static temperature

T_{4t} - total inlet temperature

T_{5t} - total exit temperature

V - volume available for combustion

V_C - volume of cylinder or basket

V_H - volume of hemisphere or dome

V_{TC} - volume of tail cone

v - velocity

v_r - reference velocity

v_4 - entrance velocity

v_5 - exit velocity

w - weight flow of air

w_f - weight flow of fuel

γ - ratio of specific heats

η - combustion efficiency

μ - viscosity of air

μ_r - reference viscosity of air

ρ - density of air

ρ_{sc} - density of air at standard conditions

APPENDIX C

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